Journal of Loss Prevention in the Process Industries 25 (2012) 391–399

Contents lists available at SciVerse ScienceDirect

Journal of Loss Prevention in the Process Industries

journal homepage: www.elsevier.com/locate/jlp

Simulation of thermally protected cylindrical container engulfed in fire

Madhusoodan Ojha [a], Amit K. Dhiman [a,*], K.C. Guha [b]

[a] Department of Chemical Engineering, Indian Institute of Technology, Roorkee, India
[b] Bhabha Atomic Research Centre, Mysore, India

ARTICLE INFO

Article history:
Received 3 June 2011
Received in revised form
24 August 2011
Accepted 6 October 2011

Keywords:
Hazardous material
Transportation
Fire
Transient heat conduction
Finite difference method
Thermal protection layer

ABSTRACT

Composite cylindrical containers are broadly used in many industries for storage and transport of hazardous materials. The focus of this study is to evaluate the performance of thermal protection layer in different fire situations likely to be encountered during an accident and to minimize the risk of loss of containment. Flame temperature is simulated based on temperature–time curve used in furnace test. The developed model for transient heat conduction in composite cylinder consisting of three layers is investigated using Crank–Nicholson finite difference method (FDM) with inclusion of temperature dependent thermal conductivity. Normal and worst case scenarios have been considered and the performance of non-ablative type thermal protection layer evaluated. Rigid foam type thermal protection layer made of different materials is considered and changes in their density (50–200 kg/m^3), conductivity (0.05–0.15 W/m K), temperature dependent conductivity and thickness of thermal protection layer (100–250 mm) on temperature profile are demonstrated for a pool fire simulating standard temperature –time curve. Variation of temperature with elapsed time at the interface between material and inner wall of the cylinder has been plotted for different thickness of protective layer (100–250 mm) which provides the vital information for preliminary assessment of protective layer thickness required to limit the temperature at the interface for the given time of exposure to fire and prevents the failure of cylindrical container.

1. Introduction

Broadly, composite cylindrical containers are used in contemporary, nuclear, aerospace, water resources and many other industries for storage and transport of hazardous materials. A recent survey on hazardous materials accidents during road transport in China from 2000 to 2008 by Yang et al. (2010) and a more comprehensive survey of road and rail accidents in the world during transport of hazardous materials of the last century till 2004 by Oggero, Darbra, Munoz, Planas, and Casal (2006) reveals the similar pattern – the most frequent accidents were releases (78%), followed by fires (28%), explosions (14%) and gas clouds (6%). It also indicates the increasing trend of the frequency of accidents. When considering the cause of major accidents, it is noteworthy that more than half are due to explosions and/or releases of hazardous chemicals (Zheng & Chen, 2011). These surveys highlight the need for continual improvement in design and other important aspects in transport to prevent the release of hazardous material after an accident. With the recent earthquake and tsunami in Japan, worst case scenario has become important for safety and risk analysis. The real fire situation depends on many factors like type of fuel, amount of fuel, geometry and area of fire, geometry and orientation of container, wind, etc.

In recent years, the effort was focused on inherently safer design of tankers, cylinders etc. used for transportation of dangerous goods/hazardous materials so as to minimize the risks. In this regard large amount of research has been done for thermal protection of LPG/petroleum transport containers which are engulfed in fire. Few other specific cases related to fire exposure along with literature covering relevant topics on heat transfer methods are considered.

Aydemir, Magapu, Sousa, and Venart (1988) developed a numerical model for LPG tanks engulfed in flames. They compared computer predictions with data from several field tests which clearly demonstrate the capability of their code in simulating the response of a cylindrical tank loaded with LPG and exposed to fire. It was one of the few which can simulate valve cycling and predict thermal stratification. Droste and Schoen (1988) reported their experimental result of fire tests on LPG storage vessels for two cases – unprotected container and same type container equipped with fire protection insulation. The unprotected tanks failed after fire duration between 7 and 12 min whereas thermal insulation was able to prevent failure even in a full fire engulfment of up to

* Corresponding author. Tel.: +91 1332 285890 (O), +91 9410329605 (M); fax: +91 1332 276535/3560.
E-mail addresses: dhimuamit@rediffmail.com, amitdfch@iitr.ernet.in (A.K. Dhiman).

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doi:10.1016/j.jlp.2011.10.001

Nomenclature

q_{net}	net heat flux, W/m^2
q_c	convective heat flux, W/m^2
q_r	radiative heat flux, W/m^2
ε_f	emissivity of flame
ε_s	emissivity of surface
ε	emissivity
F	view factor
ρ	density of material, kg/m^3
C_p	specific heat capacity, J/kg K
α	thermal diffusivity, m^2/s
r	radius, m
Δr	grid spacing, m
R_o	outer radius, m
R_i	inner radius, m
Gr	Grashof number
Pr	Prandtl number
Nu	Nusselt number
A	area, m^2
T_f	temperature of flame, °C
T_s	temperature of surface, °C
T_∞	ambient temperature, °C
T	temperature, °C
t	time of exposure, min
h_c	convective heat transfer coefficient, W/m^2 K
h_r	radiative heat transfer coefficient, W/m^2 K
h	combined heat transfer coefficient, W/m^2 K
k	thermal conductivity, W/m K
σ	Stefan–Boltzmann constant, 5.67×10^{-8} W/m^2 K^4
Δt	time increment, s
t_c	characteristic time parameter
R^*	dimensionless radius
T^*	dimensionless temperature

90 min. Beynon, Cowley, Small, and Williams (1988) also developed a model "HEATUP" to predict the behavior of containers filled with flammable liquid when they are engulfed in fire. Most recently Landucci, Molag, and Cozzani (2009) and Landucci, Molag, Reinders, and Cozzani (2009) conducted two large-scale diesel pool fire engulfment tests on LPG tanks protected with intumescing materials to test the effectiveness of thermal coatings in the prevention of hot BLEVE accidental scenarios in the road and rail transport of LPG.

Behavior of cylindrical composites when exposed to fire was studied by Henderson, Wiebelt, and Tant (1985) and developed a sophisticated model that considers the processes of heat conduction, pyrolysis, and diffusion of decomposition gases. They evaluated the accuracy of their model by comparing the theoretical temperature against measured temperature profiles for a glass/phenolic composite exposed to a high heat flux (279.9 kW/m^2) for up to 800 s.

Hydrogen is considered to be an important energy carrier in the 21st century and its application in the automotive field has become the focus of many hydrogen energy research institutions. Economic and safe hydrogen storage technology has been a critical issue. Among the several existing hydrogen storage methods, high-pressure hydrogen storage is the most favorable for long term viability. An explosion will probably occur when the high-pressure hydrogen storage vessel is subjected to a fire accident. Hu, Chen, Sundararaman, Chandrashekhara, and Chernicoff (2008) developed a non-linear finite element model for predicting the behavior of composite hydrogen storage cylinders subjected to high pressure and localized flame impingements. Whereas Ferrero, Beckmann-Kluge, Schmidtchen, and Holtappels (2010) developed model for predicting the heating-up of an acetylene cylinder involved in a fire and the afterward cooling with water are presented. Timothy Marker and Diaz (1999) evaluated the performance of oxygen cylinder overpack when exposed to elevated temperature. Oxygen cylinder overpack is designed for thermal protection that could prevent the over pressurization of cylinders during a suppressed cargo fire and the potential increase in fire hazards associated with the release of oxygen.

Transient heat transfer in composite cylinders have been extensively studied and reported in open literature. Most of them are devoted to the technique/numerical methods used for investigation of transient temperature behavior for different type of geometries, boundary conditions, composite cylinders, functionally graded materials (FGM) etc. The transient heat conduction problem in composite media was studied by Chen and Chen (1989). They presented explicit solution for the case of an infinitely long hollow cylinder composed of three different materials when subject to convective boundary conditions. Lin and Chen (1992) extended the technique of the Laplace transform to solve non-linear problems by incorporating the variation of thermal conductivity with temperature in the analysis of transient heat conduction problems. They have demonstrated their methodology for very thick multilayered cylinder that are made of highly conductive materials like metals. An analytical solution was proposed by Bairi and Laraqi (2003) for the case of fast transient conduction in spherical and cylindrical geometries subject to sudden and violent thermal effects on its surface. These solutions were presented in the form of a diagram for specific range of Bi and Fo numbers constituting a special implementation of Heisler's charts. Wang and Mai (2005) established the solution method for 1D transient temperature and thermal stress fields in non-homogeneous materials using finite difference and mode superposition techniques. Lu, Tervola, and Viljanen (2006) presented an analytical method leading to the solution of transient temperature filled in multi-dimensional composite circular cylinder with time-dependent boundary condition (periodic function) for a small range of temperature (20 K). Zheng, Su, and Su (2009) developed an improved lumped parameter model for the transient heat conduction of a wall subjected to combine convective and radiative cooling. Asgari and Akhlaghi (2009) considered a thick hollow cylinder made of two dimensional functionally graded material (2D-FGM) subjected to transient thermal boundary conditions. They combined the two methods namely finite element method with graded material properties within each element is used to model the structure and the Crank–Nicolson finite difference method is implemented to solve time-dependent equations of the heat transfer problem. Chen, Li, Shen, and Xue (2010) proposed method for investigation of axisymmetric temperature of linear heat conduction in multilayer cylinder for the case of steady state with convective boundary and demonstrated their method for large diameter furnaces like blast furnace. Tunc and Venart (1984/1985) carried out the study related to the incident radiant heat flux on the surface of horizontal cylinder which is engulfed in pool fire. They have also reported the effect of various parameters like wind direction/speed on heat flux.

Thermal protection system (TPS) is used in hypersonic reentry vehicle for protection against viscous dissipation. The vehicle surface takes on a very high temperature that even materials with high melting points cannot withstand. The ablative TPS takes the incident aerodynamic heat and either melts or sublimes away which lowers the temperature at the surface of the vehicle and decreases the effective heat flux.

Accidental exposure to fire may occur in diverse fields. Measures to tackle fire hazard depend on physical and chemical properties of hazardous material and its special characteristics. Protective clothing made of fire resistant fabrics is widely used in many industries in order to provide protection from fire. It can be modeled as composite cylinder and maximum time of exposure may be calculated for given maximum allowable skin temperature. In another case, some hazardous materials in nuclear industry are stored and transported in cylindrical containers which are protected from accidental fire by providing a thermal protection layer of suitable thickness. Thus, as far as known to us, it can be concluded that the study related to thermal protection layer for such applications is not studied in the open literature. It has motivated us to study the various aspects related to the use of thermal protection layer for the protection of cylindrical containers against accidental fire for a given period of exposure. A simplified approach to take into account of the time varying boundary condition along with temperature dependent thermal conductivity in a multilayered cylindrical container is performed for temperature distribution and the effect of various parameters is investigated. The performance of thermal protection layer (TPL) is evaluated by simulating (a) standard time–temperature curve and (b) constant flame temperature.

2. Modeling heat conduction

The cylinder containing hazardous material is covered by thermal protection layer which is generally made of rigid foam type insulating material of required thickness. A simple cross sectional view of the model and grid representation are shown in Figs. 1 and 2. To simulate the accidental fire situation, it is assumed that the cylindrical container is engulfed in fire i.e. surface is covered uniformly by flame or fire plume. The natural combustion of a horizontal fuel surface, i.e. the pool fire, is one of the most basic forms of fuel combustion often present in accidental fires. The flame temperature is typically related to the flame length which is related to the heat release rate of the fire and a characteristic length, usually the dimensional size of the burning area. The temperature within flame is reasonably constant and depends on the type of fuel which is about 800–900 °C for most types of solid fuels and the maximum temperature of 1150–1250 °C in case of hydrocarbon pool fires and natural gas (Mudan & Croce, 1995). This kind of combustion is characterized by very low-initial-momentum

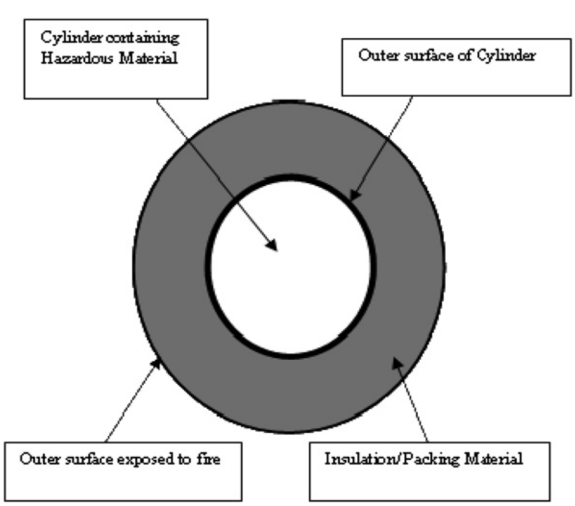

Fig. 1. Cross sectional view of the model.

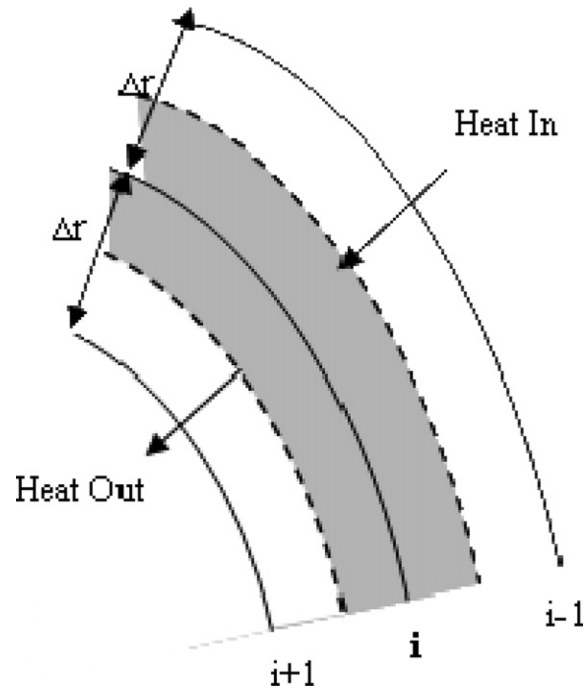

Fig. 2. Nodes for transient heat conduction in hollow cylinder.

diffusion flames and is strongly influenced by buoyancy effects. It has been well established in the literature that radiation makes a significant contribution to the overall heat transfer in flames of moderate scale (Joulain, 1996).

The accidental fire is also characterized by sooty nature of flames. The flame temperature is considered to be a function of time. In the high-temperature heat transfer case, a small amount of participating gases such as CO_2 and H_2O can bring about significant radiative heat transfer and contribute to overall heat transfer in flames (Kaminski, Fu, & Jensen, 1995). This is an important point, as the flame temperature has a great influence on the model and, thus, on the results derived from it. Considering the standard time–temperature curve to simulate the furnace test conditions for fire plume or flame temperature similar to ASTM-E-119 is calculated by equation (1) which is recommended by ISO 834-1, EN 1991-1-2 (Gardner & Ng, 2006) and IS-3809

$$T_f - T_\infty = 345 \log(8t + 1) \tag{1}$$

The relationship in equation (1) is generally referred to as the cellulosic heating rate, and although it does not correspond to an actual fire, it provides a standard basis upon which the fire performance is generally evaluated. ASTM-E-119 provides standard time–temperature values for “Standard Fire Tests” which is compared with equation (1) and shown in Fig. 3. Therefore, two kind of fire scenario can be considered:

1. Cylinder is exposed to fire from the beginning i.e. cellulosic heating rate (normal case)
2. Cylinder is suddenly engulfed in a fully developed fire (worst case scenario)

Both cases of fire are likely but worst case scenario gives us the extreme condition likely to be faced which is important from safety design point of view. Therefore, worst case scenario is also considered in this simulation study and the mean flame temperature is assumed to be ~1000 °C (simulating the hydrocarbon pool fire).

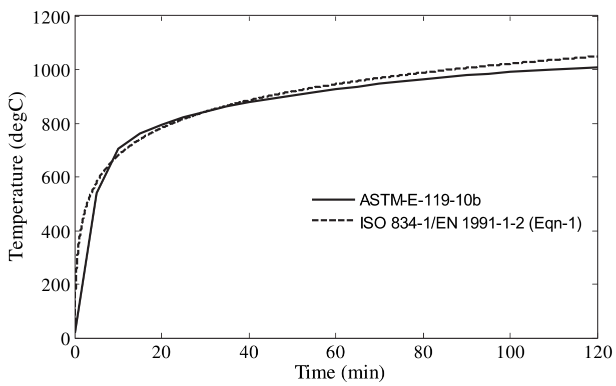

Fig. 3. Comparison of time–temperature curve recommended by different standards.

The thermal exposure from fire is typically modeled as a heat flux boundary condition. The net heat flux from a fire is composed of both convection and radiation and the net heat flux into the TPL is described by equation (2)

$$q_{\mathrm{net}} = q_{\mathrm{r}} + q_{\mathrm{c}} \tag{2}$$

And the overall heat balance at the boundary is represented by equation (3)

$$-k\frac{\partial T}{\partial r} = q_{\mathrm{net}} = \sigma F\left(\varepsilon_{\mathrm{f}} T_{\mathrm{f}}^4 - \varepsilon_{\mathrm{s}} T_{\mathrm{s}}^4\right) + h_{\mathrm{c}}\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{3}$$

The first term on the right hand side of equation (3) is the radiation from the fire, the second term is the radiative loss from the surface of cylinder, and third term is the convective heat transfer between the fire and the surface of the cylinder. The heat flux from a fire can be calculated by knowing the local gas temperature, T_{f}, surface temperature, T_{s}, the emissivity of the surface, ε_{s}, local heat transfer coefficient and other parameters. The flame which is luminous, sooty, and optically thick will result in higher emissivity of flame, ε_{f}.

The convective heat transfer coefficient is a function of the fluid properties, the flow parameters and the geometry of the surface of the heated object. It is also a function of temperature as fluid properties vary with temperature, and although convection will occur at all stages of a fire, it is particularly important at low temperatures where radiation levels are low. The empirical correlation given in equation (4) as applicable for laminar and turbulent flow, depending on the magnitude of the Rayleigh number ($\mathrm{Ra} = \mathrm{Gr}\cdot\mathrm{Pr}$) is used to calculate the convective heat transfer coefficient (Zhu, Zhang, & Song, 2008).

$$\mathrm{Nu} = 0.59(\mathrm{Gr}\cdot\mathrm{Pr})^{1/4}, \quad 10^4 < \mathrm{Ra} < 10^9 \tag{4a}$$

$$\text{and} \quad \mathrm{Nu} = 0.13(\mathrm{Gr}\cdot\mathrm{Pr})^{1/3}, \quad \mathrm{Ra} > 10^9 \tag{4b}$$

where $\mathrm{Gr} = (\Delta\rho/\rho)\,(gL^3/\nu^2)$ and $\mathrm{Pr} = C_{\mathrm{p}}\mu/k$

The variation of h_{c} with temperature is not much compared to radiation and many authors have recommended to use single value of convective heat transfer coefficient of $h_{\mathrm{c}} = 15\ \mathrm{W/m^2\,K}$ (Atkinson & Drysdale, 1992; O'Connor, Silcock, & Morris, 1996).

The emissivity of TPL is surface dependent and a reasonable range of 0.7–0.8 (sooted surface) is generally recommended for carbon steel as TPL is guarded by a thin sheet of steel. To simplify the radiative part in equation (3), it is assumed that the emissivity of flame is equal to surface emissivity i.e. $\varepsilon_{\mathrm{f}} = \varepsilon_{\mathrm{s}} = \varepsilon$. When the container is immersed completely in the plume, the exposed area is the total surface area of the container and the view factor from the effective area of the container to the fire, $F = 1$. The radiative heat transfer can be simplified after using these assumptions and the radiative part of equation (3) is written as equation (5).

$$q_{\mathrm{r}} = \sigma F\varepsilon\left(T_{\mathrm{f}}^4 - T_{\mathrm{s}}^4\right) \tag{5}$$

$$q_{\mathrm{r}} = \sigma F\varepsilon\left(T_{\mathrm{f}}^2 + T_{\mathrm{s}}^2\right)\left(T_{\mathrm{f}} + T_{\mathrm{s}}\right)\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{6}$$

$$q_{\mathrm{r}} = h_{\mathrm{r}}\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{7}$$

where $h_{\mathrm{r}} = \sigma F\varepsilon(T_{\mathrm{f}}^2 + T_{\mathrm{s}}^2)(T_{\mathrm{f}} + T_{\mathrm{s}})$

Now putting the q_{r} from equation (7) and q_{c} in equation (2)

$$q_{\mathrm{net}} = h_{\mathrm{r}}\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) + h_{\mathrm{c}}\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{8}$$

$$q_{\mathrm{net}} = (h_{\mathrm{r}} + h_{\mathrm{c}})\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) = h\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{9}$$

The equation (9) represents the simplified case of convective BC in which convective and radiative heat transfer coefficients are combined and represented as h. Therefore the boundary condition is finally represented by equation (10).

$$-k\frac{\partial T}{\partial r} = h\left(T_{\mathrm{f}} - T_{\mathrm{s}}\right) \tag{10}$$

With consideration of long cylinder and full engulfment in fire, the mode of heat transfer in the protective layer will become the 1D heat conduction in radial-direction of an insulating material that is exposed to flame/plume uniformly from all direction at the outer surface. Fire resistant rigid phenolic foam is generally used as thermal protection layer which is assumed to be non-ablative and isotropic. The outer and inner surface of the container wall is in perfect contact with the TPL and material inside the container respectively i.e. no contact resistances.

Material inside the cylinder may be in liquid or solid phase which will get heated up as the elapsed time increases. The heat transfer between wall and material inside container is by natural convection for liquid and conduction for solid material neglecting the ullage (empty space). Since the temperature gradient at the interface of cylinder and its content shall be low, for the case of cylinder with TPL, heat transferred into the material can easily be calculated using equivalent conduction method. Though container wall is thin compared to the thickness of TPL, it is still considered because of high thermal inertia. Neglecting the ullage in the cylinder, material is assumed to be uniformly distributed in the cylinder with no contact resistance at the interface. Hence complete system is treated as three layers transient conduction model with conduction as the main heat transfer mode from cylinder body to material inside cylinder even for liquid phase using equivalent thermal conductivity concept as described earlier. Since our focus is on the reduction of heat transferred into the cylinder and limit the temperature rise inside the cylinder to a safe limit by using TPL, increase in internal pressure and stress in the cylinder wall is assumed to be well within safe limit and omitted in this study.

There shall be some loss of mass at high temperature due to thermal decomposition and accompanied heat effect etc. particularly in outermost side of TPL. But it is assumed to be negligible. Therefore after assumptions like no heat generation, isotropic material, constant density, and constant heat capacity, the governing model equation for one dimension can be written for cylindrical coordinate as

$$\rho_i C_{pi}\frac{\partial T}{\partial t} = \frac{1}{r}\frac{\partial}{\partial r}\left(rk_i\frac{\partial T}{\partial r}\right) \tag{11}$$

2.1. Boundary conditions

The initial/ambient temperature can vary depending on the location and may be approximated as 40 °C which is the average ambient temperature during summer in tropical countries like India. But following the standard temperature–time curve (ASTM-E-119) initial temperature has been chosen as 20 °C. The inner boundary condition has pronounced effect on temperature distribution and internal pressure in the cylinder. Based on the above assumptions, initial and boundary conditions (IC and BC) are given below.

$$\text{IC}: T(r,0) = T_0 = 20\,^\circ\text{C}$$

$$\text{BC}-1: -k_1\frac{\partial T}{\partial r} = q_{\text{net}} = h\left(T_\text{f} - T_\text{s}\right)$$

$$\text{BC}-2: -k_3\frac{\text{d}T}{\text{d}r} = 0 \text{ at } r = 0$$

$$T(r,t) = 20\,^\circ\text{C at } r = R_\text{i} \text{ (for validation)}$$

$$\text{BC}-\text{At interfaces}: k_1\frac{\partial T}{\partial r} = k_2\frac{\partial T}{\partial r}$$

$$k_2\frac{\partial T}{\partial r} = k_3\frac{\partial T}{\partial r}$$

3. Numerical methodology

The solution of equation (11) is obtained by numerical method, but before that we have to obtain the outer boundary condition, i.e., transient surface temperature. The outer surface temperature of the TPL varies with flame temperature and time. The flame temperature variation with time as per equation (1) is used to simulate the furnace test condition as per the standards. The surface temperature rise depends on the flame temperature as well as thermal properties of the material. Since the conductivity and density of TPL material is generally low resulting low thermal inertia of TPL material and quick increase in surface temperature is expected. Surface temperature is calculated separately by applying the boundary condition taking the energy balance across the first half of the outer element. The net rate of heat in by convection and radiation − the rate of heat out by conduction = the rate of heat accumulation in Δt, s:

$$\rho_1 C_{p1}\left(A\frac{\Delta r}{2}\right)\frac{\partial T}{\partial t} = hA\left(T_\text{f} - T_\text{s}\right) - k_1 A\frac{\partial T}{\partial r} \tag{12}$$

Since the grid spacing is very small, the variation in area in equation (12) is neglected. The equation (12) is solved for T_s (which is T_1 for first nodal point) using finite difference scheme and final equation is given below after simplifying assumptions and rearrangement (13)

$$T_\text{s}^{j+1} = \frac{2\alpha h\Delta t}{k}T_\text{f}^j + \left[1 - \left(\frac{2\alpha h\Delta t}{k\Delta r}\right) - \frac{2\alpha\Delta t}{\Delta r^2}\right]T_\text{s}^j + \frac{2\alpha\Delta t}{\Delta r^2}T_2^j \tag{13}$$

The equation (13) is used to calculate the surface temperature by explicit method which is subsequently used as boundary condition to solve the main equation (11).

There are various approaches to solve the partial differential equation (11). In case of constant thermal conductivity, analytical solution is available for single layer. But for transient temperature distribution in multilayer cylindrical body as well as accounting for variation in thermal conductivity with temperature analytical solution is not available, hence suitable numerical method is used for solving equation (11). In this study, Crank–Nicolson (C–N) form of implicit approach of finite difference method (FDM) has been used to develop the numerical code for solving the PDEs. The composite cylindrical system has three layers – first layer consist of thermal protection layer, second layer is thick metallic wall of the cylindrical container and third layer consist of material inside cylinder. The material inside cylinder may be in solid or liquid phase. To avoid the convective boundary for liquid phase, effective thermal conductivity has been used based on equivalent thermal conductivity concept. The equation (11) is converted into algebraic equation using C–N method and simplified equation is written below.

$$\begin{aligned}\frac{k\left(T_{i-1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}T_{i-1}^{j+1} - \left[1 + \frac{k\left(T_i^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2}\right]T_i^{j+1} + \frac{k\left(T_{i+1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}T_{i+1}^{j+1} \\ = -T_i^j - \frac{k\left(T_{i+1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}T_{i+1}^j + \frac{k\left(T_i^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2}T_i^j \\ - \frac{k\left(T_{i-1}^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2}T_{i-1}^j\end{aligned} \tag{14}$$

The equation (14) is a linear algebraic equation when thermal properties are constant. For incorporating temperature dependent thermal conductivity and still avoiding non-linearity, the thermal conductivity is evaluated at temperature calculated in previous time level at the corresponding spatial coordinate which is known and used in equation (14). The equation (14) is represented as

$$A_{i-1}T_{i-1}^{j+1} - B_{i-1}T_i^{j+1} + C_{i-1} = D_{i-1} \tag{15}$$

Where

$$A_{i-1} = \frac{k\left(T_{i-1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}$$

$$B_{i-1} = 1 + \frac{k\left(T_i^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2} \quad C_{i-1} = \frac{k\left(T_{i+1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}T_{i+1}^j$$

$$D_{i-1} = -T_i^j - \frac{k\left(T_{i+1}^j\right)\Delta t}{2\rho C_\text{p}(\Delta r)^2}T_{i+1}^j + \frac{k\left(T_i^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2}T_i^j - \frac{k\left(T_{i-1}^j\right)\Delta t}{\rho C_\text{p}(\Delta r)^2}T_{i-1}^j$$

The above equation (15) provides Tridiagonal system matrix when written for all spatial nodal points and can be solved simultaneously, after applying suitable boundary conditions, using Thomas algorithm. It has been applied for composite or multilayer cylinder by suitably changing the thermal properties for the given thickness of material.

4. Results and discussion

The numerical code for the solution of above model is developed in MATLAB. Initially grid independence checked and found to be satisfactory with $\Delta r \leq 0.005$ m with error less than 0.5%. Reducing the element size and thereby increasing the number of nodal points usually yields a more accurate solution but at the cost of an increased computing time and memory requirement. In this study

fine grid spacing of $\leq$0.002 m (i.e. 0.008 in dimensionless term) is used with adequate time interval (<1 s).

4.1. Surface temperature

The surface temperature of TPL depends on the flame temperature, combined heat transfer coefficient and thermal properties of TPL material. Since thermal diffusivity of insulating material is very low, the outer surface temperature will rise very quickly and will be very close to flame temperature within short duration. The effect of time dependent and constant flame temperature on surface temperature are shown in Fig. 4 considering the same value of combined heat transfer coefficient (HTC) and thermal properties. In both the cases surface temperature is close to flame temperature for $h = 50$ W/m^2 K. Combined HTC has significant effect on surface temperature which is shown in Figs. 5a and b. For the same value of h, surface temperature also depends on the thermal properties of the material of TPL which has also been demonstrated and shown in Fig. 5c by varying the thermal conductivity of material from 0.05 to 5.0 W/m K. Though inclusion of radiation effect allows further higher value of combined HTC, conservatively $h = 50$ W/m^2 K has been chosen here as the common value for all the subsequent calculations as it gives the surface temperature close to flame temperature.

4.2. Model validation

Validation of result was carried out by considering constant conductivity and constant surface temperature using characteristic time parameter, $t_c = \rho C_p(R_o - R_i)/k$, and compared with the results of Wang and Mai (2005) for homogeneous material, with temperature profile at $t = 0.01t_c$, $0.1t_c$, $0.2t_c$ and $0.3t_c$. Wang and Mai (2005) state that "The transient response is obtained via mode superposition, which gives a solution with only errors introduced by the spatial discretization. Their Fig. 2 depicts the temperature distribution at $t = 0.01\tau_0$; $0.1\tau_0$ and $0.3\tau_0$; where $\tau_0 = \rho c(b-a)^2/k$ is a characteristic time parameter. Also depicted is the steady-state solution. The results are in excellent agreement with the series solutions." But there is a discrepancy in graph shown by Wang and Mai (2005). For instance, the curve stated as $t = 0.1t_0$ is actually for $t = 0.2t_0$ and curve for $t = 0.1t_0$ is not shown in that figure. Therefore, we have included all these curves in this study, as shown in Fig. 6. The dimensionless temperature values extracted from the plots of Wang and Mai (2005) are given in Table 1 with corresponding values calculated here by us using C–N method for time $t = 0.3t_0$ and steady state. An excellent agreement exists between the two studies.

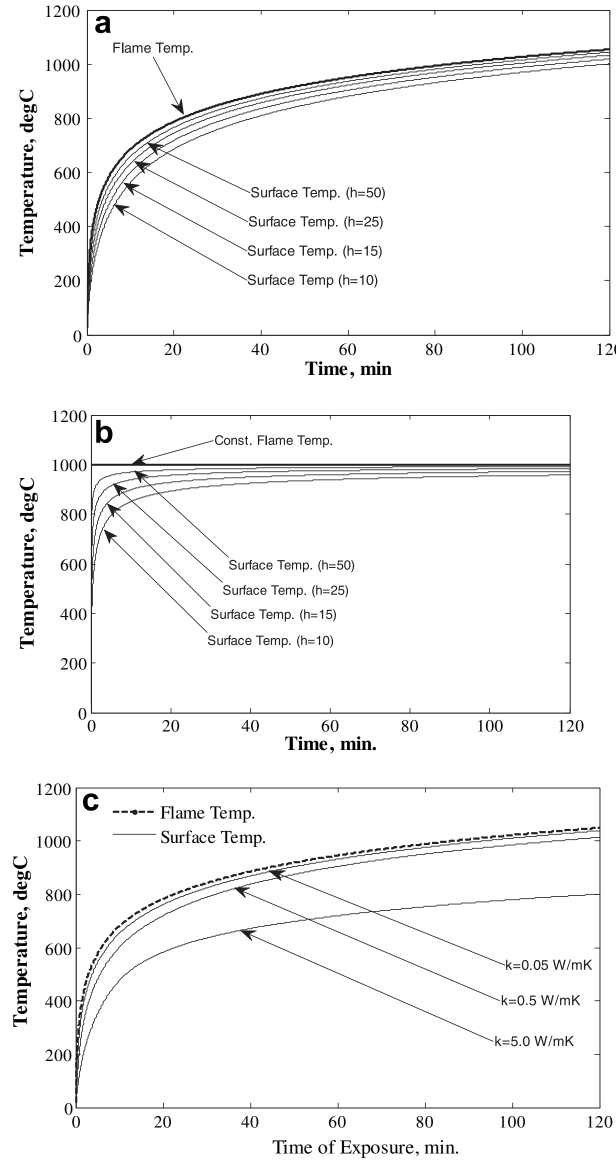

Fig. 5. (a) Comparison of surface temperature for different values of combined HTC for varying flame temperature. (b) Comparison of surface temperature for different values of combined HTC for constant flame temperature. (c) Comparison of surface temperature for different values of conductivity ($h = 50$ W/m^2 K).

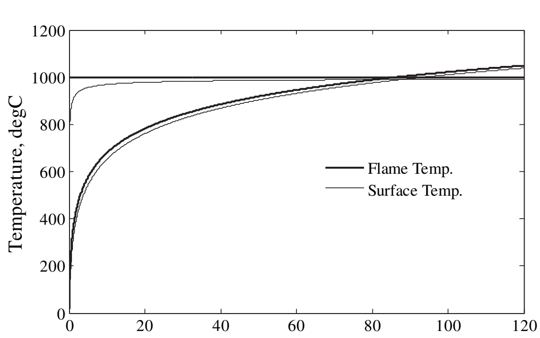

Fig. 4. Comparison of flame temperature for constant and time varying flame temperature ($h = 50$ W/m^2 K).

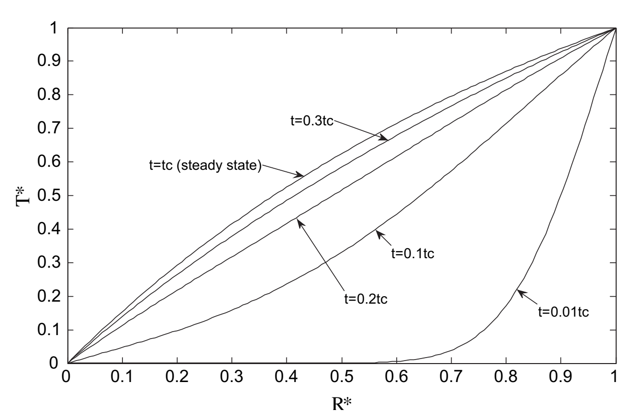

Fig. 6. Temperature profile in homogeneous hollow cylinder extended up to steady state (SS) based on characteristic time parameter(t_c).

Table 1
Validation of present results with the results of Wang and Mai (2005).

Radius	Temperature at time $t = 0.3t_c$		Temperature at time $t = t_c$ (steady state)	
	Present work	Wang and Mai (2005) (Interpolated approx. value from graph)	Present work	Wang and Mai (2005) (Interpolated approx. value from graph)
1	1	1	1	1
0.92	0.942	0.94	0.951	0.95
0.84	0.882	0.87	0.898	0.89
0.76	0.817	0.81	0.841	0.85
0.68	0.75	0.74	0.78	0.8
0.6	0.678	0.68	0.715	0.72
0.52	0.604	0.6	0.643	0.65
0.44	0.526	0.53	0.566	0.57
0.36	0.444	0.44	0.483	0.49
0.28	0.357	0.36	0.391	0.4
0.2	0.265	0.28	0.292	0.31
0.12	0.165	0.17	0.183	0.2
0.04	0.058	0.07	0.064	0.07
0	0	0	0	0

Effects of changes in various thermal properties of insulating materials and boundary conditions can be visualized using the code. The temperature profiles for validation have been plotted in dimensionless form i.e. dimensionless temperature (T^*) and dimensionless radius of insulation (R^*) including container wall thickness. These dimensionless temperature and radius are calculated from corresponding temperature (T) and radius (r) as

$$T^* = \frac{T - T_0}{T_s - T_0} \quad R^* = \frac{r - R_i}{R_o - R_i}$$

The dimensions of cylindrical container, wall thickness, insulation material and its thickness are approximately selected based on commonly used size of the cylinder in industry. The thermal properties of materials (rigid foam, container wall and material inside the container) are to be referred in Table 2 wherever it is not mentioned specifically.

4.3. Performance evaluation

The performance of TPL is evaluated by simulating the variations of different thermal and physical properties of TPL material. The transient temperature distribution in the rigid foam and container is determined at different time of exposure (10, 30, 60 and 120 min) and shown in Fig. 7. The temperature rise at the interface between material and cylinder wall after 60 min of the fire engulfment is negligible for the given thickness of 235 mm and thermal properties of material. Effect of flame temperature on inside temperature distribution has been shown in Fig. 8 for normal and worst case scenarios after 60 min of exposure. As the time of exposure increases effect of flame temperature subsides and approaches to normal case. Therefore worst case scenario is important when time of exposure to be considered is less than 60 min. Effects of various parameters on temperature distribution are discussed below.

Table 2
Thermal properties of steel, rigid foam and material inside cylinder.

	Material contained in cylinder	Carbon steel as container wall (Chen & Chen, 1989)	Rigid foam type insulation material
Density, kg/m^3	2000	7833	100
Heat capacity, J/kg K	1000	465	1000
Thermal conductivity, W/m K	0.1	54	0.05
Radius/thickness, mm	400	15	235

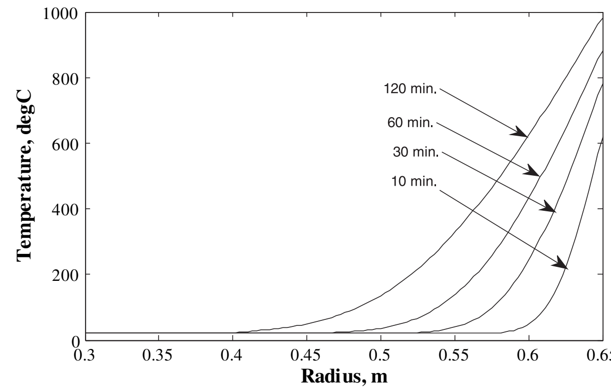

Fig. 7. Temperature distribution inside thermally protected cylinder at different time of exposure.

4.3.1. Different material for thermal protection layer (TPL)

Thermal conductivity of various insulating materials differs widely which has significant effect on temperature distribution and one of the most important criteria for the selection of material for TPL. Various values of thermal conductivity are used to simulate different kind of material for TPL and temperature profiles are shown in Fig. 9 after 60 min of the fire engulfment. The duration of 60 min fire exposure is chosen based on the assumption that within this time fire may subside or fire personnel will be able to reach the accident site and control the fire. If local laws already stipulate fire test duration then same elapsed time shall be used for such analysis. For the given thickness of TPL the maximum temperature rise with high conductivity (0.15 W/m K) is just nominal from the initial temperature of the system, keeping density and heat capacity of the TPL material constant.

4.3.2. Variation of conductivity with temperature

In case of high-temperature environment material properties like density, specific heat and thermal conductivity etc. generally becomes temperature dependent. Variation in density and specific heat with temperature may be negligible but thermal conductivity of rigid foam can vary according to the density, operating temperature and type of gas entrapped within the voids of the foam. Generally thermal conductivity increases with temperature for foam type insulating material. The air or gas within the voids becomes more excited as its temperature is raised, and this

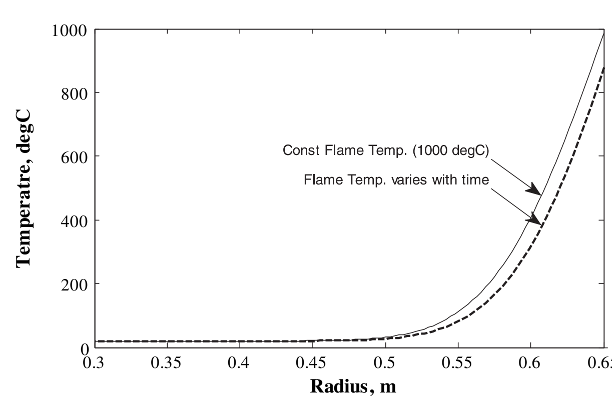

Fig. 8. Comparison of temperature distribution after 60 min of exposure for cases of normal and worst scenarios.

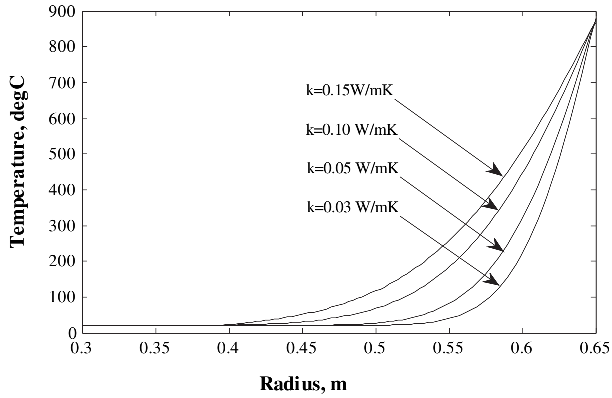

Fig. 9. Temperature profile in different TPL material: effect of thermal conductivity (temperature profile after 60 min of exposure).

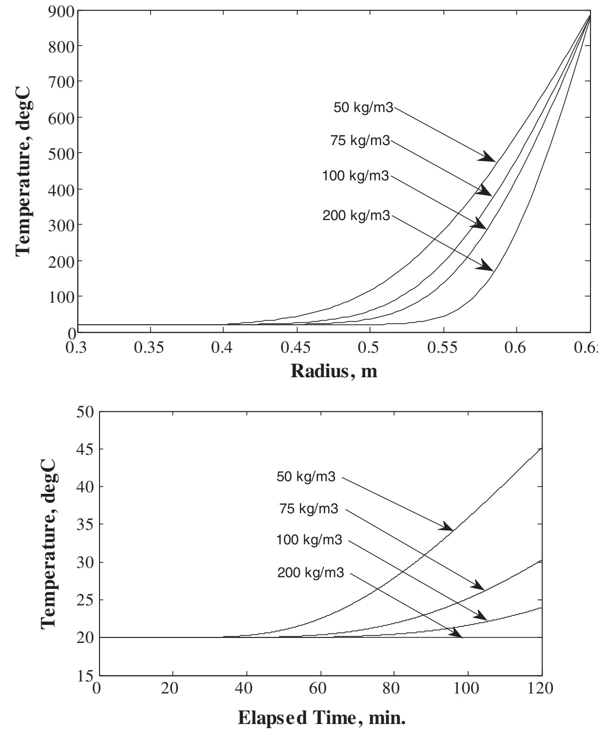

Fig. 11. (a) Effect of density of different TPL material: temperature profile after 60 min of exposure. (b) Effect of density of different TPL material: interface temperature variations with time of exposure.

enhances convection within or between the voids, thus increasing heat flow. This increase in thermal conductivity is generally continuous for air-filled products and can be mathematically modeled.

Variation of thermal conductivity has been incorporated in the model equation by evaluating it at the previous time step for avoiding the non-linear term (Wang & Mai, 2005) which simplifies the calculation with negligible error as time step needed is very low due to stability criteria of C–N method. The thermal conductivity of phenolic and other rigid foam type insulation materials are available in literature at room temperature. Since close cell structure of foam contains entrapped blowing gas during foam production, actual relationship between temperature and thermal conductivity of given insulation should be determined experimentally, as it is a combination of conduction and convection in the rigid foam. Microstructure of foam shall provide the idea about open and closed cell structure which is very important for temperature dependency of the conductivity. Due to lack of reliable data, it is assumed that conductivity is linearly dependent on temperature similar to air and following relationship is used in this study.

$$k = 0.05 + 0.00015T \text{ W/m}\,^{\circ}\text{C}$$

The above relationship is incorporated in the code and temperature profile is determined for 60 min of exposure which is shown in Fig. 10a and compared with constant conductivity. The

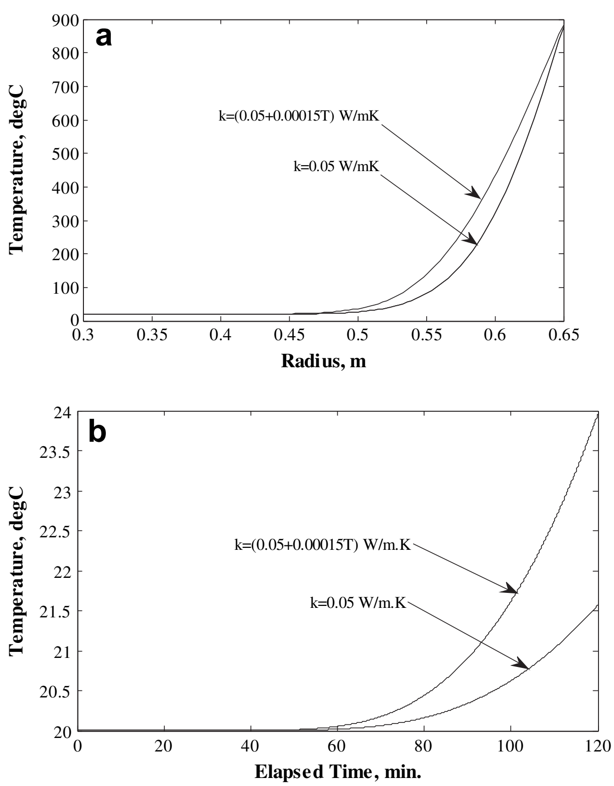

Fig. 10. (a) Effect of variation of thermal conductivity with temperature: temperature profile after 60 min of exposure, $k(T) = 0.05 + 0.00015T$. (b) Effect of variation of thermal conductivity with temperature: Interface temperature vs. elapsed time, $k(T) = 0.05 + 0.00015T$.

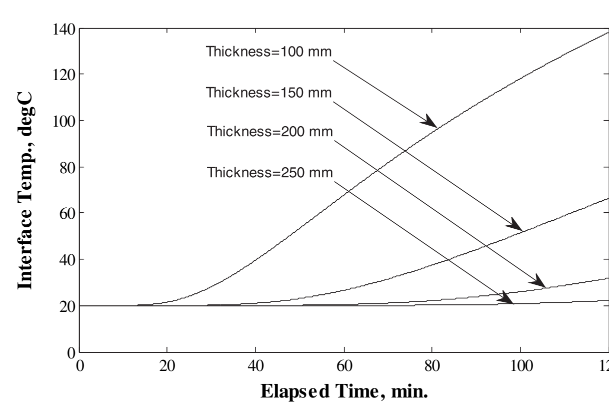

Fig. 12. Temperature profile at the interface between material and cylinder wall with respect to elapsed time: effect of thickness of TPL.

comparison of interface temperature variation for constant and temperature dependent conductivity is shown in Fig. 10b. Comparing the temperature distribution with constant conductivity (0.05 W/m K), it shows the approximately 3.5 °C temperature rise after 60 min of exposure when thermal conductivity variation with temperature is considered.

4.3.3. Density and thickness of thermal protection layer

Density plays an important role in rigid foam type insulation material which is generally low and ranges from 50 to 200 kg/m^3. But for special applications where sufficient strength is also required the density of rigid foam can be as high as 500 kg/m^3. The effect of density on temperature distribution, keeping same value of conductivity, is plotted after 60 min of exposure to fire and shown in Figs. 11a and b. The increase in density also increases volumetric heat capacity, thermal inertia, and resulting decrease in the temperature rise in the TPL material in transient conditions while assuming constant conductivity. The interface temperature increases by 21 °C when density decreases from 100 to 50 kg/m^3 which may significantly increase the vapor pressure depending on the characteristic of the material inside the cylinder.

The effect of change in thickness of TPL on interface temperature with elapsed time is depicted in Fig. 12 which indicates that the thickness of 150 mm is suitable to limit the interface temperature below 65 °C for 2 h of exposure and 100 mm for 1 h of exposure. It provides the most important data related to the required thickness of TPL for the given set of thermal properties of rigid foam and time of exposure to fire. It emphasizes the need for optimization for given set of parameters and requirements.

5. Conclusions

In this paper, simulation study is performed for the temperature distribution in a thermally protected cylindrical container which was exposed to pool fire at ∼1000 °C. One-dimensional transient heat conduction problem is solved for multilayer cylinder using Crank–Nicholson finite difference method. Temperature dependent property like thermal conductivity has been incorporated in the code. The results are validated by simulating to single layer and comparing well with literature data. Simulation study has been carried out for two cases – normal and worst cases. Increase in density of rigid foam from 50 to 200 kg/m^3 has decreased the temperature at the interface significantly. But increase in density is generally accompanied with increase in the thermal conductivity which shall nullify the effect of increase in density. The temperature at the interface has increased from 21.5 to 24 °C after 60 min of exposure to fire when temperature dependent thermal conductivity is considered. The temperature at the interface has increased about 110 °C for decrease in thickness of TPL from 250 to 100 mm. This study provides the valuable insight into the effect of various thermal properties as well as physical dimension on temperature profile inside the insulating material for different time of exposure which helps in the optimization of thickness of thermal protection layer.

Acknowledgments

The authors would like to express their gratitude to Shri T K Bera and Shri S Sarkar, BARC Mysore, for their guidance and support during this study.

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